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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

The Prediction of Bearing Performance

PUBLISHED BY
THE TEXAS COMPANY
TEXACO PETROLEUM PRODUCTS

Bearing Research at Pennsylvania State College

PREDICTION of bearing performance is of the utmost value to the designing engineer, in planning for bearing construction which will be commensurate with operating requirements.

In the realization of this fact The Texas Company has sponsored a program of bearing research at the Pennsylvania State College, which has developed some very pertinent and practical data.

The transposition of theoretical research data to practical application though very often a difficult undertaking, has been successfully accomplished in the cooperative studies with the research personnel at Penn State. The following discussion and actual illustrations of its workability through typical problems is presented with the feeling that it will be of distinct interest to all who are concerned with bearing performance and effective lubrication.

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The Prediction of Bearing Performance

Study of the Problems Involved

THE determination of data in the interest of predicting proper bearing performance prior to construction has been the basis for considerable research work at The Pennsylvania State College*, under the sponsorship of The Texas Company. In particular this work has been conducted with the purpose in mind of developing means for practical application of earlier investigations involving theoretical pressure distribution, coefficient of friction and journal running position. With these data available, it is felt that more intelligent selection of lubricating oils to meet existing bearing design is practicable.

In studying the distribution of pressure within an oil film when the shaft is completely separated from the bearing by the film, and the ascertainment of the running position of the shaft with respect to the bearing, the first step was to determine the load at which metallic contact between the shaft and bearing occurred. This was done by placing a 3-volt flashlight in a circuit which included the oil film, and so insulating the machine that a complete circuit was impossible unless the shaft and bearing were in electrical contact. It was found that the light began to glow when the load reached a value of 415 to 425 lbs. per sq. in. of projected area, for a speed of 500 r.p.m. Since the light did not glow at pressures less than these it was concluded that the shaft and bearing were completely separated when the pressure was less than 415 lbs. per sq. in.

Two values of speed were originally used, 500 and 750 r.p.m. In later tests the machine was run at 500, 1600, 2100 and 3000 r.p.m. with loads of 100, 200, 300 and 400 lbs. per sq. in. of projected bearing area at each speed.

All this research has pertained to sleeve type, end lubricated bearings of one length-to-diameter ratio, with a view to development of simplified curves and formulae, which can be more readily applied than much of the research data, which has heretofore been available in this connection.

The intensity with which the results have been studied and verified by actual experiment over a wide range of speed, load and oil viscosity justify decided dependability upon the substantiated mathematical basis, particularly as no experimental disagreement has, as yet, been discovered within the field.

The clearance to diameter ratio was also varied and it is one of the important features of this work that this effect can be taken care of in the use of a single curve, having the proper coordinates, the justification for which is shown in the subsequent mathematical work under the heading "Journal Running Position."

By reason of the outstanding simplicity and the probable value of the results to designing and operating engineers, it is felt that a discus-

* Under the direction of Prof. L. J. Bradford, by E. M. Barber and C. C. Davenport. See paper "Bearing Characteristic Curves for Fluid Film Lubricated Bearings," by L. J. Bradford and C. C. Davenport, presented before the American Society of Refrigerating Engineers, December 7-9, 1932.

sion of the manner in which the research has been handled and its application to certain practical problems, will be of distinct interest. Particular attention has been given to the analysis of those factors with which such engineers will be confronted. These involve:

(1) The minimum practicable oil film thickness which can be allowed in operation.
(2) The arc of contact of the supporting oil film.
(3) Friction with respect to heat generating capacity and its effect upon power consumption.

In this study the work has been directed along two different lines, (1) An extension of the investigations of oil film pressure distribution under varying conditions of load, speed and bearing clearance, with a view to determining the effect of pressure distribution on correct bearing design and on the load carrying capacity of the bearing, and (2) the development of an interrelation among the many variables encountered in problems of bearing design and operation, so that a complete solution for such problems can be obtained.

Minimum Film Thickness

In consideration of this factor it has been fully realized that bearing failure will occur only when there is actual contact between the metals of the journal and the bearing or, in other words, when boundary lubrication has failed.

The minimum thickness of the oil film is, therefore, the obvious factor in determining the extent to which effective bearing operation can be depended upon.

In view of the wide variation in bearing practice it is practically impossible arbitrarily to assign definite values of safe minimum film thickness. This value must be chosen with due respect to the factors involved.

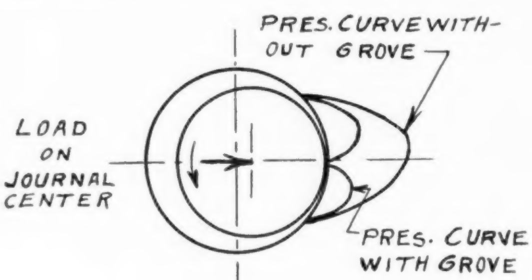


Fig. 1—Curve illustrating the effect produced by placing an oil groove in the load carrying region of the bearing.

(1) Even with very careful machining small bumps and irregularities are to be found on the surface of a bearing. The magnitude of these irregularities will, of course, be dependent upon the care and precision with which the bearing surface was prepared. The minimum film thickness must be greater than such variations in the bearing surface.

(2) Shaft deflection within the bearing must also be taken into consideration and the minimum film thickness must cover this effect in addition to surface irregularities.

(3) An excess of minimum film thickness sufficient to cover possible variations of speed, load and temperature must also be allowed.

Arc of Contact of Supporting Film

In order to insure satisfactory results in bearing lubrication the active portion of the bearing or, in other words, that area over which positive film pressures are essential, must be provided with a copious and uninterrupted supply of oil.

Remembering that the carrying capacity of a bearing is proportional to the area under the film pressure curve, Fig. 1 clearly illustrates the effect of a discontinuity in the film such as would be produced by placing an oil groove in this region.

The importance of definite knowledge concerning the extent of the high pressure region is at once recognizable and has accordingly been chosen as a factor of major importance.

Friction

Friction, the quantity which determines the power loss and heat generating capacity of the bearing, is obviously of considerable importance. It is of further importance in that it serves as a very convenient check on the journal running position for so long as the power lost and heat generated lie within certain ranges, fluid film lubrication prevails and there is no contact between the metallic surfaces.

Theoretically and actually two frictions have been determined; (1) bearing friction and (2) journal friction.

(1) Bearing friction is the measure of the torque required to prevent the rotation of the bearing with the shaft. It is the friction generally determined by experiment but has no great practical significance.
(2) Journal friction is the resistance to the rotation of the shaft and is the friction in which we are actually interested, for it is the true measure of the power loss and heat generating capacity of the bearing.

Description of the Apparatus

The machine used in this work was described in the May-June issue of **"Lubrication"**, 1931, with the exception that slight modifications have been made in the bearing shell in order to provide a positive oil pressure on both

ends of the bearing, and thus eliminate air leakage into the oil space on the non-active side of the bearing.

Briefly, the machine consists of a journal, approximately 2½ inches in diameter, supported on two self-aligning bearings. Between the two bearings and around the journal is a bearing block lined with babbitt. Variation of the shaft diameter is .0001 inch and of the bearing .0003 inch. Near the ends of this bearing shell two comparatively large oil grooves are arranged circumferentially around the shell, one on each end, the purpose of which, as stated above, is to keep a positive oil pressure on the bearing at all times and prevent air leakage. Between these two side grooves the shell is drilled with five radial holes along the length of the bearing. These radial holes connect with four axial holes which in turn are provided with pressure gauges. This arrangement allows the measurement of the actual pressure in the oil film at five different positions along

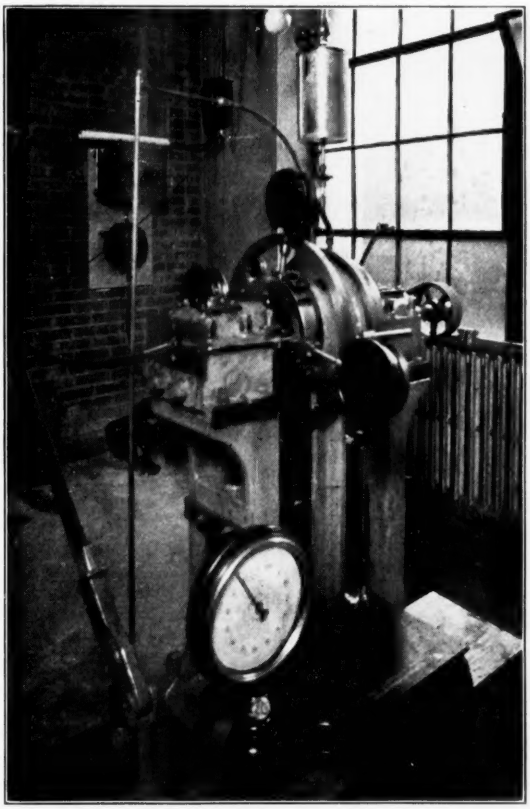


Fig. 2—General view of the machine used in the bearing research investigations at Pennsylvania State College.

the axis of the bearing. The curves in Figure 4 are a typical illustration of the data obtained for a load of 200 pounds per square inch and a speed of 500 r.p.m.

In order to get the distribution of pressure around the journal at these five different longitudinal positions the bearing block was made movable in an outside steel yoke. In this manner, the direction of pressure on the block could be kept constant and the measuring holes rotated around the journal in respect to this direction of pressure. When it was desired to move the bearing shell inside of the yoke, an oil was injected temporarily between the two and the shell was moved easily.

The relative position between the axis of the journal and the axis of the shell under different loads was determined by means of accurate gauges at both ends of the bearing. The friction was determined by the tendency of the whole bearing assembly to be turned around the shaft at different speeds. The pressure was maintained by means of the deflection of a steel bar between the journal supports and the bearing block. This bearing is calibrated so that the actual pressure corresponding to a specific deflection is known. There are several other details in order to eliminate errors and counter-balance undesirable forces. Full information can be obtained if desired from the laboratory at the Pennsylvania State College.

METHOD AND SCOPE OF INVESTIGATION

The experimental work which was essential for the obtaining of data preparatory to development of the necessary charts and formulae was carried out on the Kingsbury journal testing machine*, as described above, due to its suitability for the determination of

(1) Journal running position.
(2) Oil film pressure distribution and,
(3) Friction observations.

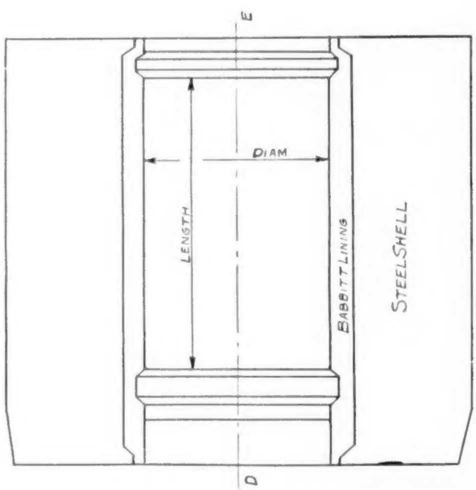
By means of the holes drilled in the surface of the bearing shell and connections to pressure gauges, determination of oil film pressure distribution is made practicable. From these data the extent of the high pressure or load supporting area of the bearing has been obtained.

By means of a group of micrometer dial gauges on each end of the bearing it has also been practicable to determine the journal running position for any given set of conditions. A brass ring attached to the bearing and connected through an arm to a scale pan enables the operator to weigh the bearing friction and later, by means of suitable calculation, to transfer this to journal friction.

Under the conditions of speed and load heretofore mentioned, using Texaco Regal Oil C of 324 seconds Saybolt Universal viscosity, at 100 degrees Fahr., for the majority of the tests, data was developed with respect to running position film pressure distribution and friction,

* For further details see also Bulletin No. 39 of the Engineering Experimental Station of the Pennsylvania State College.

and a correlation developed between the dynamic and geometrical variables entering the problem.



	BRNG 4	BRNG 5	BRNG 6
MEAN DIAM.	2.4996	2.5014	2.5047
LENGTH	3.875	3.875	3.875
CLEARANCE	.0046	.0064	.0097

Fig. 3—Longitudinal section of the bearing, showing the large oil grooves on each end.

SYMBOLS EMPLOYED

a = Radius of journal in inches
a + n = Radius of bearing in inches
n = Radial clearance in inches
C = Eccentricity factor
A = Angle to leading edge of supporting film
B = Angle to trailing edge of supporting film
Z = Absolute viscosity centipoises
N = Revolutions per minute
h = Minimum oil film thickness
L = Axial length of bearing
D = Nominal diameter of bearing
U = Surface velocity of journal = $\frac{2\pi aN}{60}$
P = Load in pounds per sq. in. of projected bearing area
R = Load in pounds per unit length of bearing
α = Attitude angle of journal
f and λ = Coefficient of friction on journal
f and λ^1 = Coefficient of friction on bearing

Journal Running Position

An understanding of the mathematical work involved in this and the two sections following is not essential to the use of the charts (Figs. 7, 8, and 10) in the solution of bearing design problems. The mathematical work merely justifies, theoretically, the existence of the sort of curves which have been developed experimentally.

In developing this correlation, equation (15) given by W. J. Harrison* for the unit carrying capacity of a bearing of infinite length was appropriately manipulated so as to yield a relationship which can be used as the coordinates for a chart on which journal running position for any conditions of operation of a complete bearing having a given length-to-diameter ratio can be plotted on a single curve.

Equation (15) given by Harrison:

$$R = \frac{12\pi ZUa^2}{n^2} \times 69 \times 10^3 \frac{C}{(2+C^2)\,(1-C^2)^{1/2}} \quad (15)$$

where $\frac{C}{(2+C^2)(1-C^2)^{1/2}}$ = a function of $C = \rho\, C$

$$R = 2\,Pa$$

$$U = \frac{2\pi a\,N}{60}$$

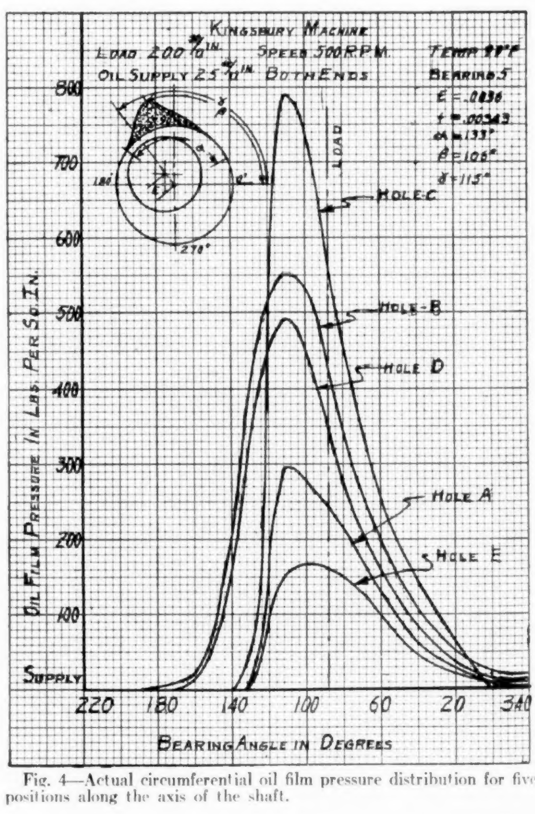


Fig. 4—Actual circumferential oil film pressure distribution for five positions along the axis of the shaft.

*The Hydrodynamical Theory of Lubrication, Cambridge Philosophical Society, Vol. XXII (1913).

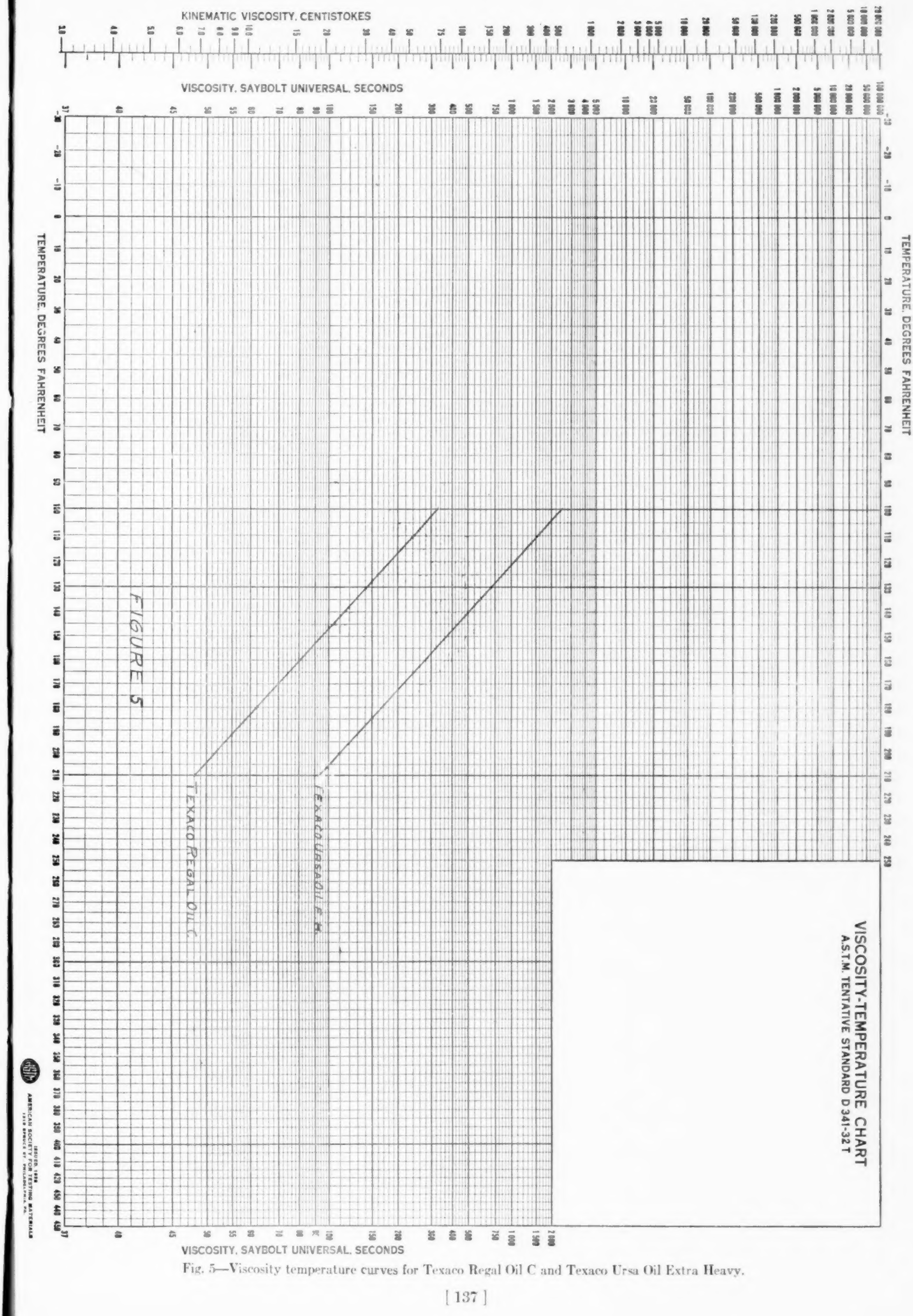


Fig. 5—Viscosity temperature curves for Texaco Regal Oil C and Texaco Ursa Oil Extra Heavy.

Substituting in (15)

$$2\ \mathrm{Pa} = \frac{12\pi^2 2 Z N a^3}{n^2 \times 60} \times 69 \times 10^3\ \rho\ C \qquad (15a)$$

Transposing

$$\rho\ C = \frac{12\pi^2 2 Z N a^3 \times 69 \times 10^3}{n^2 60\ 2\mathrm{Pa}} \qquad (15b)$$

Removing the constant terms

$$C \sim \frac{ZN}{P} \left(\frac{a}{n}\right)^2 \qquad (15c)$$

The relationship (15c) is the desired correlation between the running position as expressed by the eccentricity factor C, and the quantities which affect it. The curve is shown in Fig. 7.

Arc of Contact of Supporting Film

In the derivation of equation (15) Harrison uses an equation of the form:

$$R = \frac{6ZUa^2}{n^2} \int_{\theta = B}^{\theta = A} (\text{Function } \theta)$$

Where A is the leading angle of the active film and B is the trailing angle by a manipulation similar to that employed in determining the relationship for the journal running position, the coordinates for the leading and trailing angle of the film can be determined as:

$$A \text{ and } B \backsim \frac{Z\,N}{P} \left(\frac{a}{n}\right)^2$$

Figure 8 shows a plot of the values of angles A and B for a wide range of operating conditions.

Bearing and Journal Friction

Harrison's Equations for bearing friction indicate that the bearing friction should have some finite value when the shaft and journal are concentric and should drop off to zero when there is metallic contact between the bearing and journal. This, however, is quite contrary to all experimental evidence not only at the Pennsylvania State College, but by other experimenters. Consequently, the relationship for the correlation of bearing friction did not hold. After trying numerous combinations, a relationship for bearing friction which, although it appears to have no justification, holds true throughout the range covered by the experiment.

$$\lambda^1 = \frac{ZN}{P} \left(\frac{a}{n}\right)$$

Figure 9 shows the plot of bearing friction against $\frac{ZN}{P} \left(\frac{a}{n}\right)$ These data are of little practical use except in the calculation of journal friction.

Although Harrison's Equations for bearing friction do not produce satisfactory results, his analysis of journal friction is much more satisfactory and coordinates may be derived from his equations 15a and 16:

$$\lambda = \frac{n}{a} \frac{(1 + 2C^2)}{3C} = \frac{n}{a} (fC) \qquad (16)$$

$$R = \frac{12\pi^2\ 2\ Z\ a^3\ N}{n^2 \times 60}\ \rho\ C \times 69 \times 10^3 \qquad (15a)$$

By proper manipulation and substituting:

$$\lambda \frac{a}{n} \sim \frac{ZN}{P} \left(\frac{a}{n}\right)^2$$

The apparatus employed in this investigation did not permit the operator actually to "weigh" the journal friction, so it was calculated indirectly. Considered statically the forces acting on the bearing shaft are equal, hence the moments must balance. By setting up the proper equation and substituting, we find

$$\lambda = \lambda^1 + C \left(\frac{n}{a}\right) \operatorname{Sin} (\alpha - 90)$$

The terms entering the calculation are all direct measurements and the results are in good accord with theory and direct weighing of journal friction.

Figure 10 shows the plot of journal friction for the operating range covered by the experiments.

Application of these data to actual problems which might confront the designing or operating engineer is indicated hereafter.

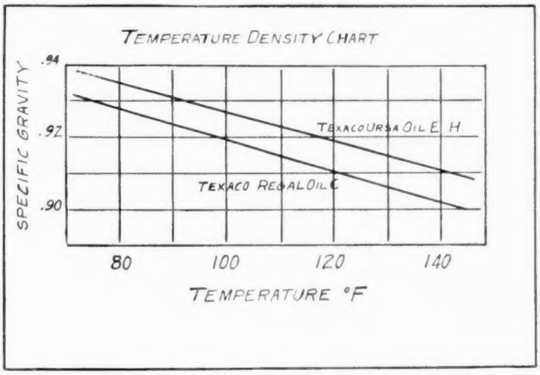


Fig. 6—Density temperature curve for Texaco Regal Oil C and Texaco Ursa Oil Extra Heavy.

EXAMPLES

I. A bearing is to carry a load of 5000 pounds when rotating at 1500 r.p.m. The allowable bearing pressure is to be 250 pounds per sq. in. and the running temperature 130 degrees Fahr. The clearance of the bearing is to be .002 inches per inch of diameter and the L/D ratio 1.55.*

Determine the length and diameter of the bearing, the power loss and the journal running position if an oil of 200 seconds Saybolt Uni-

* The L/D ratio of the bearing investigated was 1.55; therefore, bearings calculated by this method are limited to these geometrical proportions.

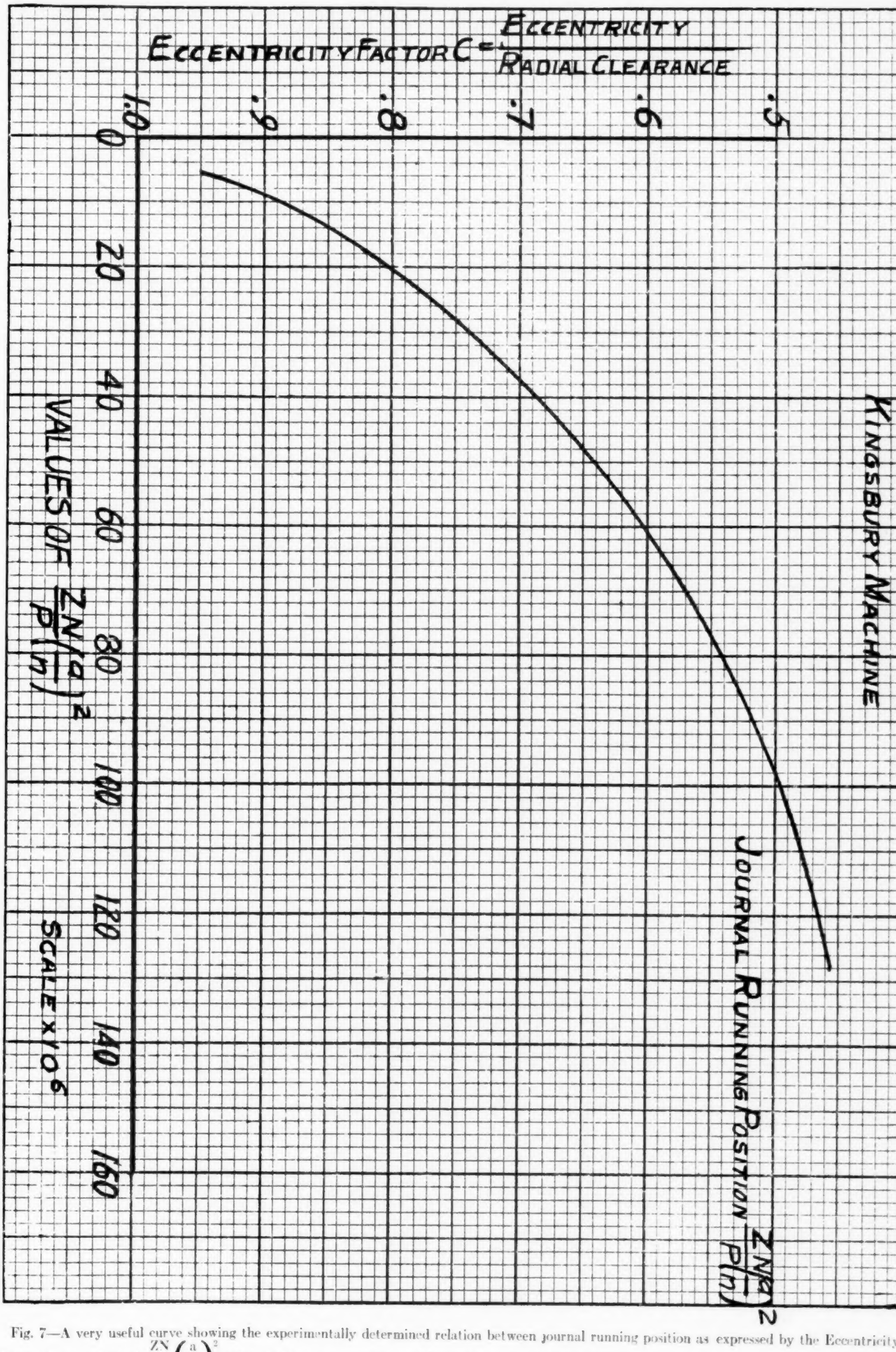


Fig. 7—A very useful curve showing the experimentally determined relation between journal running position as expressed by the Eccentricity Factor C and values of $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ as justified in the paragraph on "Journal Running Position."

versal viscosity and a specific gravity of 0.92 at the running temperature is used.
Bearing area required:

$$\text{Projected Area} = \frac{\text{Total Load}}{\text{Unit Load}}$$
$$= \frac{5000}{250} = 20 \text{ sq. in.}$$
$$= L \times D$$
$$L = \frac{L}{D} \text{ ratio} \times D = 1.55\ D$$

Substituting
Projected Area $= 1.55\ D^2$
Transposing and taking the square root

$$D = \sqrt{\frac{\text{Projected Area}}{1.55}}$$
$$D = \sqrt{\frac{20}{1.55}} = \underline{3.6} \text{ in.}$$
$$L = \frac{L}{D} \text{ ratio} \times \text{Diameter}$$
$$L = 1.55 \times 3.6 = \underline{\overline{5.58}} \text{ in.}$$

Radial Clearance $n = \frac{D}{2} \times$ bearing clearance
Per inch of D.

$$n = \frac{3.6}{2} \times .002$$
$$n = .0036 \text{ in.}$$

To determine the power loss, we know

$$\frac{\text{Radius of Journal } a}{\text{Radial Clearance } n} = \frac{1.8}{.0036}$$
$$\frac{a}{n} = 500$$

The absolute viscosity in centipoises Z can be determined from the Viscosity Temperature Chart,* Figure 5 and Temperature Density Chart, Fig. 6.

$$Z = 42 \times .92 = 38.7 \text{ centipoises}$$
$$C \backsim \frac{ZN}{P}\left(\frac{a}{n}\right)^2 \qquad (15c)$$
$$\frac{ZN}{P}\left(\frac{a}{n}\right)^2 = \frac{38.7 \times 1500 \times 500^2}{250}$$
$$= 58 \times 10^6$$

From Figure 7 we determine from the curve that $C = .607$

The minimum film thickness h is equal to the radial clearance minus the eccentricity factor times the radial clearance.
That is:
Minimum film thickness h

$$= n - C\,n = n\,(1 - C)$$
$$= .0036\,(1 - .607)$$
$$= .0036 \times .393 = .00142 \text{ inches}$$

To determine the friction refer to the curve in Figure 10 and for a value of 58×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ we find:

* The A.S.T.M. Viscosity-Temperature chart, issued 1932 by American Society for Testing Materials, Tentative Standard D341-32T, 1315 Spruce St., Phila., Penna., can be used.

$\lambda\left(\frac{a}{n}\right)$ to be 3.16

Therefore, the coefficient of friction on the journal

$$\lambda = \frac{3.16}{\left(\frac{a}{n}\right)} = \frac{3.16}{500} = .0063$$
$$\text{Power Loss} = \frac{2\pi N T}{33000 \times 12}$$
$$= \frac{2\pi \times 1500 \times 5000 \times .0063 \times 1.8}{33000 \times 12}$$

Power Loss $= 1.35$ H.P.

II. What friction and minimum film thickness will result from changing the oil used in Example I to an oil of 100 seconds Saybolt Universal viscosity and a specific gravity of 0.92 at the running temperature.

From the Viscosity Temperature Chart, Figure 5

$$Z = 19.5 \times .92 = 17.9 \text{ centipoises}$$
$$\text{Then } \frac{ZN}{P}\left(\frac{a}{n}\right)^2 = \frac{17.9 \times 1500 \times (500)^2}{250}$$
$$= 26.8 \times 10^6$$

From the curve in Figure 7 we find that $C = .762$

Minimum film thickness $h = n\,(1 - C)$.

$$= .0036\,(1 - .762)$$
$$= .00086 \text{ in.}$$

To determine the coefficient of friction on the journal refer to Fig. 10 and we find from the curve that

$\lambda\left(\frac{a}{n}\right) = 1.88$ for the value of 26.8×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$

$$\lambda = \frac{1.88}{\left(\frac{a}{n}\right)} = \frac{1.88}{500} = .00376$$

III. A bearing having a diameter of $2\frac{1}{4}$ inches and an L/D ratio of 1.55 carries a load of 100 pounds per sq. in. of projected area. The journal rotates at 2000 r.p.m. Measurements show that the difference between the diameter of the bearing and that of the journal is .0075 inches. It is desired to carry the load with a minimum film thickness of .0003 inches. What should be the viscosity of the oil at the running temperature and what H.P. loss in friction may be expected?

Length of bearing $L = 1.55 \times 2.25 = 3.5$ inches.

Area $= L \times D = 2.25 \times 3.5 = 7.88$ sq. inches.

Total Load $=$ Area $\times$ load per sq. inch.
$= 7.88 \times 100 = 788$ lbs.

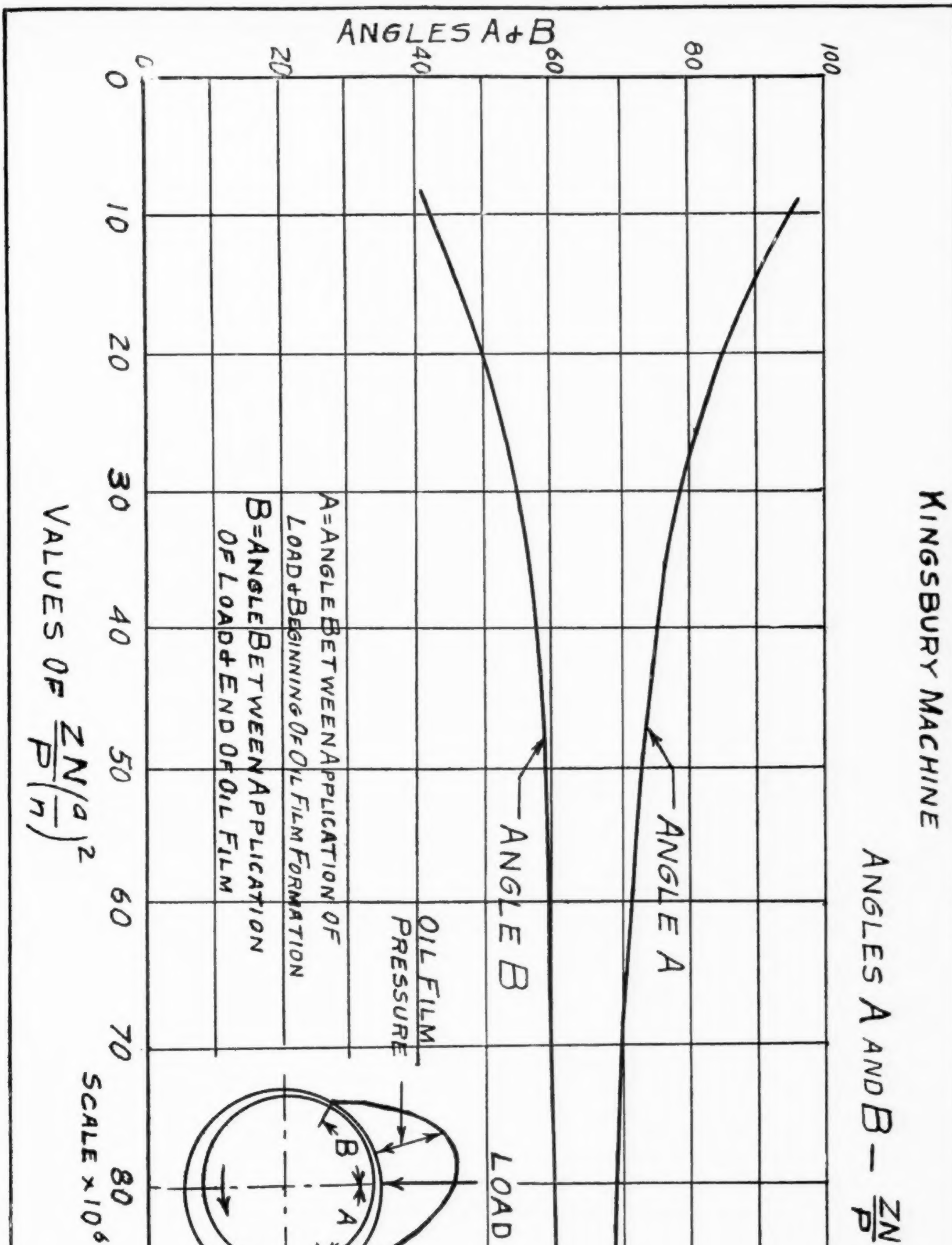


Fig. 8—A very useful curve showing the experimentally determined relation between the values of $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ and the angles of positive oil film pressure as justified in the paragraph on Arc of Contact of Supporting Film.

$$\text{Radial Clearance } n = \frac{.0075}{2} = .00375 \text{ inches.}$$

The minimum film thickness h is equal to the radial clearance n minus the eccentricity factor C times the radial clearance n.

$$h = n - Cn$$

$$\text{or} \quad C = 1 - \frac{h}{n} = 1 - \frac{.0003}{.00375} = .92$$

From the curve in Figure 7 we find a value of 7.9×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ for an eccentricity factor C of .92.

$$\frac{a}{n} = \frac{1.125}{.00375} = 300$$

$$\frac{ZN}{P}\left(\frac{a}{n}\right)^2 = 7.9 \times 10^6$$

$$Z = \frac{7.9 \times 10^6 \times P}{\left(\frac{a}{n}\right) \times N} = \frac{7.9 \times 10^6 \times 100}{300^2 \times 2000}$$

$$= 44 \text{ centipoises}$$

If we wish the running temperature to be 120 degrees Fahr., and know the oil to be used, has a specific gravity of .91 at this temperature, the Saybolt Universal viscosity of the oil can be determined from the Viscosity Temperature Chart, Figure 5.

$$\text{Kinematic Viscosity} = \frac{\text{Centipoises}}{\text{Specific Gravity}} = \frac{44}{.91}$$

$$= 48.3$$

From the Viscosity Temperature Chart the Saybolt Universal viscosity of the oil is 225 seconds at 120 degrees Fahr. To determine the

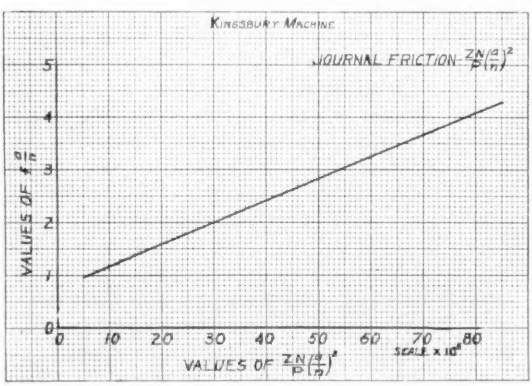


Fig. 9—Curve showing the bearing friction.

power loss, refer to Figure 10 and from the curve for a value of 7.9×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ we find a value of 1.1 for $\lambda\left(\frac{a}{n}\right)$

$$\lambda = \frac{1.1}{300} = .00367$$

$$\text{Power loss} = \frac{2\pi NT}{12 \times 33{,}000}$$

$$= \frac{2\pi\, 2000 \times 788 \times .00367 \times 1.125}{12 \times 33{,}000}$$

$$= .112 \text{ HP}$$

IV. In the case of the bearing derived in Example I determine location and extent of the region that should be free from grooves.

In Example II we found a value of 26.8×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ when using the 100 seconds Saybolt Universal viscosity oil at the running temperature.

Refer to Figure 8 and from the curves we find that Angle A is equal to 82 degrees and Angle B to be 53 degrees for the value of 26.8×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$

The region covered by these arcs is the load supporting area of the bearing and should be adequately supplied with oil and be free from discontinuities.

V. Assuming the conditions to remain the same as in Example II, determine the effect of machining the shaft to a tolerance of $\pm$ .001 inches.

The radial clearance n = .0031 to .0041.

$$\frac{a}{n} = \frac{1.8}{.0031} \text{ to } \frac{1.8}{.0041}$$

$$= 581 \text{ to } 440$$

$$\frac{ZN}{P}\left(\frac{a}{n}\right)^2 = 36.3 \times 10^6 \text{ to } 20.8 \times 10^6$$

From Figure 7 the eccentricity factor C for the values of 36.3×10^6 and 20.8×10^6 for $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ will be

$$C = .702 \text{ to } .800$$

The minimum film thickness

$$h = n\,(1 - C)$$

$$h = .000925 \text{ inches to } .00082 \text{ inches.}$$

The limits of coefficient of journal friction can be determined by referring to Fig. 10, where we find values of 2.28 to 1.63 for $\lambda\frac{a}{n}$

Substituting 300 for $\frac{a}{n}$ and solving

$$\lambda = .00392 \text{ to } .00371$$

The examples illustrate the possibilities of solving bearing problems both from the standpoint of the bearing designer and the lubrication engineer. At present only one bearing has been investigated using a number of speeds,

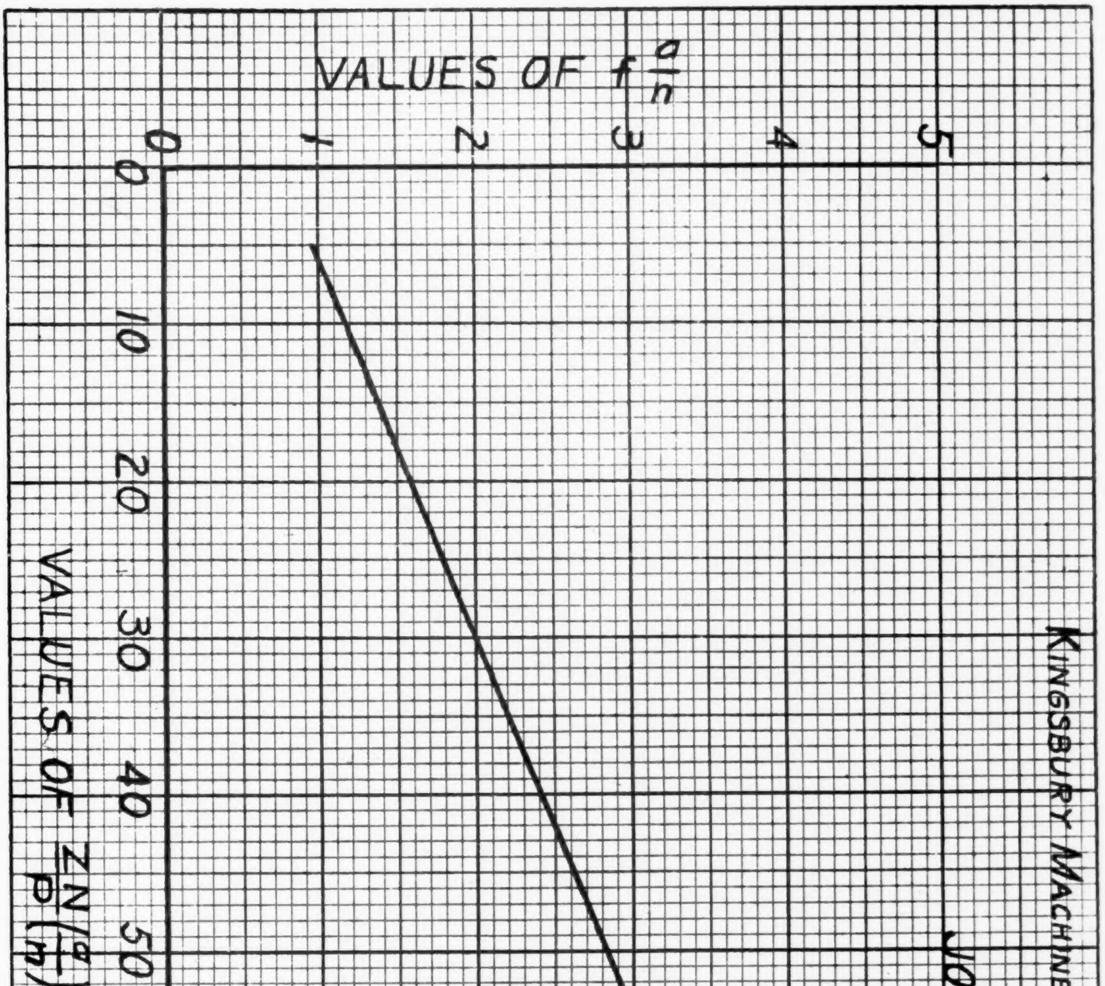


Fig. 10—A very useful curve showing the experimentally determined relation between the values of $\frac{ZN}{P}\left(\frac{a}{n}\right)^2$ and values of f $\left(\frac{a}{n}\right)$ from which f, the coefficient of journal friction can be determined as discussed in paragraph on "Bearing and Journal Friction."

pressures, lubricating oils, temperatures, and clearances, but as the length-to-diameter ratio of this bearing was 1.55, bearings calculated using the data and curves in this issue of "LUBRICATION" are limited to this ratio of 1.55. It will be a very simple matter to extend this investigation to cover other L/D ratios.

Conclusion

The studies described in the foregoing pages are distinctive of the interest which is being shown in the application of the theory of lubrication to practical machine design and operation. It is indicative of the realization that effective bearing lubrication is an economic necessity, dictated by the present day high standards of engineering efficiency and competition. For proper lubrication, after all, means longer life, lower lubricant consumption, lower costs of maintenance, and repair, and higher efficiency in power transmission.

Studied in conjunction with such factors as load, operating speed, bearing clearance, means of application and the physical characteristics of lubricants, data as developed by the various authorities referred to in this article can be used to excellent advantage, if the practical side of the question is given due consideration. One must never lose sight of the fact, however, that the ultimate objective of the engineering personnel of any plant is to keep machinery running, to insure constant and maximum production, whatever the product. Research data should, therefore, be so developed as to further this program. It will be all the more appreciated by the practical operator in this regard and a source of satisfaction to the research student in the knowledge that his efforts have been directly applied to the furtherance of our worldwide needs for economical production.

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